Empirical Lumped-mass Approach to Modelling Heat Transfer in Automotive Turbochargers

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Abstract

When evaluating the performance of new boosting hardware, it is a challenge to isolate the heat transfer effects inherent within measured turbine and compressor efficiencies. This work documents the construction of a lumped mass turbocharger model in the MatLab Simulink environment capable of predicting turbine and compressor metal and gas outlet temperatures based on measured or simulated inlet conditions.

A production turbocharger from a representative 2.2L common rail diesel engine was instrumented to enable accurate gas and wall temperature measurements to be recorded under a variety of engine operating conditions. Initially steady-state testing was undertaken across the engine speed and load range in order that empirical Reynolds-Nusselt heat transfer relationships could be derived and incorporated into the model. Steady state model predictions were validated against further experimental data.

Model predictions for compressor wall temperature show very good correlation with measured data (average 0.4% error, standard deviation 1.27%) and turbine housing temperatures also demonstrate good agreement (average 2.7% error, standard deviation 3.58%). The maximum compressor and turbine wall temperature errors were 2.9% and 8.1% respectively at the steady state validation test conditions.

This work demonstrates that a relatively simple approach to modelling the heat transfer within turbochargers can generate accurate predictions of housing temperatures and the knock-on impact on compressor and turbine gas outlet temperatures.

Introduction

Shayler et al. [1] outlined the fundamental heat transfer calculations which treated the sections as lumped masses with an associated temperature as well as assuming quasi-steady flow conditions within the pipe. The pipe section itself is treated on the basis of turbulent convective heat transfer from the gas to the pipe inner wall, conductive transfer radially through the wall and axially along the section and by natural convection to the surroundings from the pipe outer wall. An energy balance is then struck between the energy loss from the gas flow and energy dissipated to the element's surrounding via conduction, convection and radiation.

Total heat transfer coefficients can be obtained using the 'resistor analogy' discussed and presented by Eriksson [2] whereby the modes of heat transfer are considered as resistors positioned in series or in parallel with the sum of the reciprocals equating to the total heat transfer coefficient as given in Equation 1.

$$\frac{1}{h_{tot}} = \frac{A_i}{A_e}\frac{1}{h_{cv,i}} + \frac{1}{h_{cd}} + \frac{1}{h_{cv,e} + h_{cd,e} + h_{rad}}$$

Equation 1 – Total heat transfer coefficient calculated using the 'resistor analogy' [2]. Where A is the surface area, h is the heat transfer coefficient and the subscripts *cv*, *cd* and *rad* represent convection, conduction and radiation respectively. Subscripts *i* and *e* denote internal and external.

ESTIMATION OF HEAT TRANSFER – For all exhaust sections an estimate of heat transfer at the inner and outer wall is required. The heat transfer calculations contained within the Matlab-Simulink model were originally obtained from published literature. As may be expected, the relationships reported by different authors varied significantly depending on the specific system being modelled.

The common way of describing the convective heat transfer relationship between the gas and inner wall of the pipe, of a given system, is by examining the correlation between the Reynolds number (Re) of the gas and the Nusselt number (Nu). The Nusselt number is a dimensionless number used to measure the enhancement of heat transfer due to convection and can be expressed for a simple pipe section as:

$$Nu = \frac{h_{cv,i} D_i}{k_f}$$

Equation 2 – Nusselt Number Relationship. Where $h_{cv,i}$ is the internal convective heat transfer coefficient, D_i is the internal diameter of the pipe section and k_f is the thermal conductivity of the fluid.

As the Nusselt number is a measure of the increased heat transfer due to convection it follows that its value will be related to the state of the fluid flow to which the surface is exposed and thus, the Reynolds number. The general form of the Re/Nu relationship is given in Equation 3.

$$Nu = c_0 \operatorname{Re}^{c1} \operatorname{Pr}^{c2} \left(\frac{\mu_{bulk}}{\mu_{skin}} \right)^{c3}$$

Equation 3 - General form of Re/Nu Relationship

Where c_0, c_1, c_2 and c_3 are experimentally derived coefficients and μ denotes the fluid viscosity.

Experimental Setup

All experimental work was undertaken on a production Honeywell VGT turbocharger fitted to a 2.2L Common rail Diesel engine installed on a transient engine dynamometer at the University of Bath. The engine itself was instrumented for pressures and temperatures in various locations within the oil and coolant circuits along with ambient air temperatures and at locations within the air intake and exhaust systems. Although the engine is of EURO VI technology level, the catalytic convertor was removed to prevent thermal interactions with the turbocharger**.**

Table 1 - Engine data

Parameter	Value
Engine Configuration	I4, Turbocharged, Intercooled
Displacement (litres)	2.2
Bore (mm)	86
Stroke (mm)	94.6
Con Rod Length (mm)	155
Compression Ratio	15.5:1
Max Speed (RPM)	4900
Max Torque (Nm)	385
Max Cylinder Pressure (bar)	160
Turbocharger	Production, Honeywell VGT
Max Turbo Speed (kRPM)	213
Max Pre-Turbine Temperature (°C)	830
Max Compressor Outlet Temperature (°C)	180

Mass air flow (MAF) into the engine was measured using an ABB Sensyflow FMT700 series hot film anemometer and exhaust flowrate calculated from MAF and instantaneous fuel flow measured using a gravimetric fuel balance.

Instrumentation

The turbocharger itself was instrumented with more than 30 thermocouples to allow the measurement of gas, oil and metal temperatures throughout the hardware. Although low thermal inertia thermocouples were considered, such as 0.5mm exposed junctions, due to reliability and robustness considerations, all measurements were taken using sheathed 1.5mm k-type thermocouples. At the entry and exit to the compressor and turbine, it was necessary to determine if there was uniform radial temperature distribution of the gas, therefore 3 thermocouples were fitted protruding into the flow at lengths of 10mm and 15mm as well as in the centre of the duct. An example of the 3 measurement locations at the turbine outlet location is given in Figure 1.

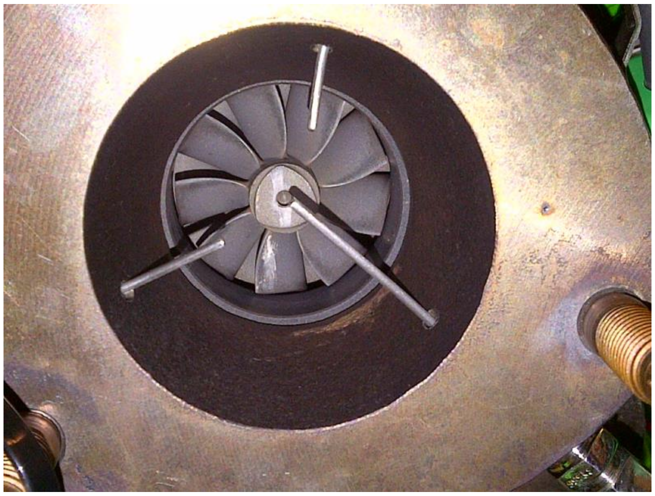

Figure 1 – Thermocouple locations at turbine outlet at the centre of the duct and at protrusions of 10 and 15mm.

Metal temperatures were measured at 3 locations around both the scroll of the compressor and turbine and, at each location, 2 thermocouples were mounted at different depths within the wall, a 'deep' measurement close to the internal wall and an outer surface temperature. Additional metal surface temperatures were measured on the bearing housing between the compressor and turbine as close as possible to the respective back plates to allow determination of the thermal gradient between turbine and compressor.

Oil temperature was measured in the feed to the bearing and in the drain together with flowrate using a gear type positive displacement flow meter.

Gas pressures were measured at the inlet and exit of the compressor and turbine using remotely located Druck transducers accessing the gas stream via pressure tappings.

Steady State Conditions

A range of steady state and transient experiments were conducted as described below.

Steady state experiments were conducted at fixed engine speeds and loads as shown in Figure 2. Prior to each measurement being taken, the engine was allowed to thermally soak for 8 minutes to allow thermal equilibrium to be attained. At each operating condition, after stabilization, data was averaged from 100 logs recorded at 1Hz. High frequency data was recorded simultaneously for 300 cycles to provide further information if required.

Data points were not collected at limiting torque and other high exhaust temperature conditions to aid with longevity of instrumentation over the extended experimental programme. However, the points examined provided a broad range of pre-turbine exhaust gas temperatures for use in model validation.

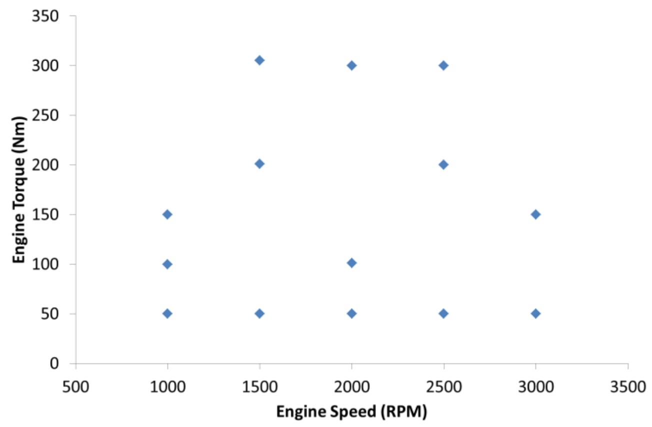

Figure 2 - Steady State experimental test points

Turbine and Model Setup

Using a similar approach to other studies presented by the author [3,4,5], the turbocharger heat transfer model utilises an empirical Re/Nu relationship derived by the University of Bath. This relationship predicts the Nusselt number based on the Reynolds number and pipe geometry data such as wall thickness and bend radius. Due to the complexity of the internal geometry of a turbine or compressor, the assumption is made that the gas is passing through a pipe section of equal internal surface area to the turbine or compressor, with the necessary data being obtained from CAD drawings of the hardware. Mass of the turbocharger components were also obtained from drawings.

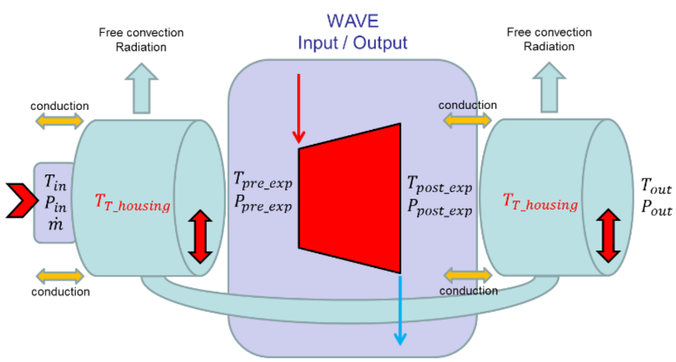

Figure 3 - Turbine model heat transfer overview

The basic structure of the developed heat transfer model for the turbine is shown in Figure 3. The turbine is broken down into three sections, an inlet duct, the expansion process and an outlet duct. Engine data from the WAVE combustion model can be used as an input providing exhaust gas temperature, pressure and flow rate information. The temperature differential between the exhaust gas and the wall then influences the heat transfer, with the exhaust gas losing energy to the turbine housing (assuming the gas is at a higher temperature to the wall) leading to a reduction in gas temperature and an increase in the housing temperature. A revised gas temperature and pressure, T_{pre_exp} and P_{pre_exp}, are now used as the input to an isentropic expansion process with the turbine efficiency included through interrogation of maps obtained experimentally on a gas stand. This is used to predict the gas temperature and pressure after the expansion at that engine operating condition which is then subject to further heat transfer within the turbine exit duct. It is important to note that all energy flow within the pre and post expansion ducts flow to a common turbine housing, and not two separate masses connected by conduction. This will often lead to minimal heat transfer in the post-expansion outlet duct as the lowered gas temperature is likely to be similar to the housing temperature which has been raised by the higher inlet duct gas temperatures. Under some engine operating conditions or transient events, there can be energy flow from the housing to the exhaust gas in the exit duct leading to a rise in the gas temperature leaving the turbine compared to the ideal post-expansion temperature.

For model validation purposes, experimental engine data was used as inputs into the model (inlet gas temperature/pressure and exhaust flow) and a simple isentropic expansion was used to represent the expansion within the turbine taking into account the efficiency of the turbine using manufacturer's turbine efficiency maps. T_{out} and $T_{T\_Housing}$ (Figure 3) were then compared with the measured exit gas temperature and an average measured housing temperature to assess the accuracy of the model.

A more simplified model was originally assessed whereby all heat transfer is assumed to occur either before or after the expansion event in the turbine, or compression event in the compressor. This simplification however did not allow energy flow in both directions between the gas and the wall and did not therefore realistically represent the actual system.

Turbocharger Model Arrangement

In order to represent the complete turbocharger system, the individual compressor and turbine models (in the configuration shown in Figure 3) were linked via conduction through a cast iron shaft of representative cross-sectional area as shown in Figure 4. The functionality to allow heat transfer from the connecting shaft to the lubricating oil was also included in the model.

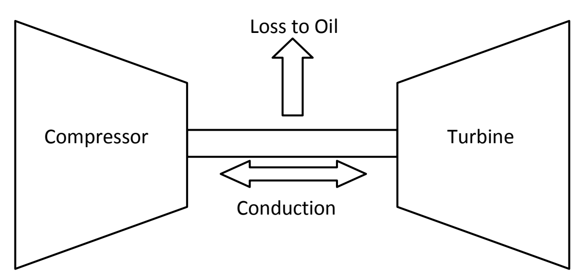

Figure 4 - Turbine / Compressor model integration

Initially the heat loss to the oil was neglected in order to simplify the modelling, however this resulted in overestimates of the compressor wall temperatures due to higher than expected energy flow from the turbine to the compressor.

As the WAVE model does not predict oil temperatures, experimental data was used to estimate a relationship between engine power and the turbo oil feed inlet temperature. The obtained relationship is shown in Figure 5. Using this estimated oil temperature, the specific heat capacity for the oil is obtained from lookup tables [6].

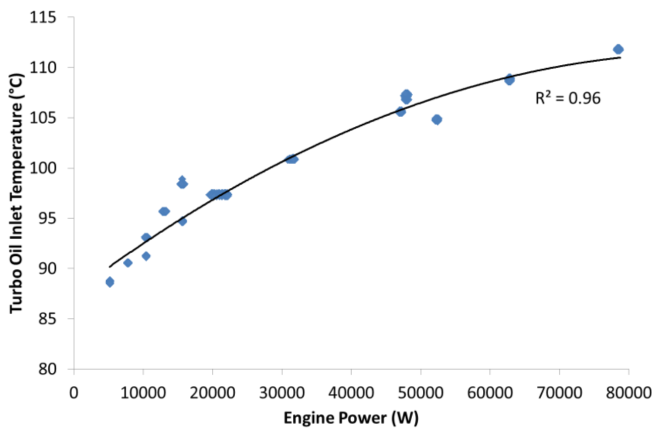

Figure 5 - Experimentally derived turbocharger oil feed inlet temperature against engine power

Unfortunately the extremely low and non-steady flow of the oil drain from the bearings produced inaccurate readings from the flow meter making the theoretical modelling of the heat transfer to the oil from the bearings impossible. Therefore, steady state experimental data was also used to derive an engine-specific relationship between engine power and the increase in oil temperature across the turbocharger bearings (Figure 6).

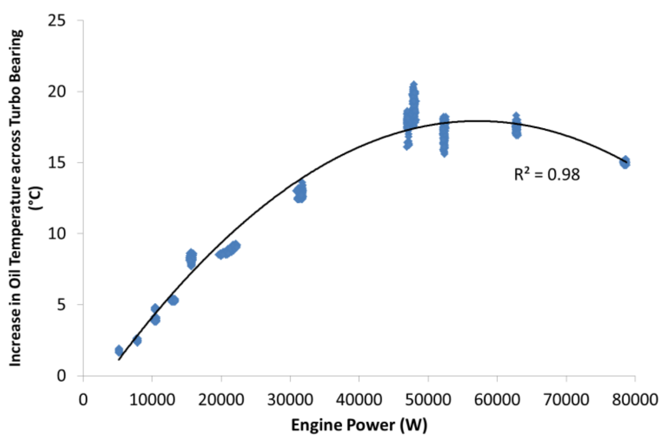

Figure 6 - Experimentally derived increase in turbocharger oil temperature against engine power

The energy required to raise the oil by the estimated dT is then calculated based on the properties of the oil [6] and subtracted from the energy flow from the turbine to the compressor via conduction to calculate the net energy flow across the bearing housing.

There are several limitations of this approach in estimating the energy flow to the engine oil in the turbocharger bearings:

- Relationships are based on steady state data and will therefore exhibit errors under cold start conditions where the engine oil has not yet reached its normal operating temperature.
- The relationships are likely to be highly hardware-specific and not transferable between engine platforms or turbocharger hardware.

Reynolds / Nusselt Relationship

The convective heat transfer coefficient is calculated using the Nusselt number which will vary with the Reynolds number of the fluid. Initially, published Re/Nu relationships were used within the model in both the compressor and turbine. However, these relationships were derived from experimental measurements of simple straight and bent, uniform diameter pipe sections and, as such, did not accurately represent the complex internal turbocharger geometry. For this reason, steady state experimental data was used to derive a new Re/Nu relationship for this hardware.

The baseline turbocharger model was used with a PID controller to determine, at each engine condition, the required Nusselt number to match the calculated and measured compressor wall temperature. Results for the Reynolds / Nusselt relationship determination are shown in Figure 7.

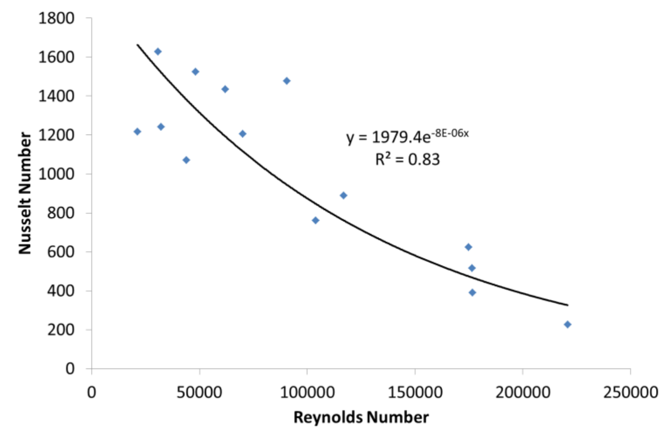

Figure 7 - Derived Reynolds / Nusselt relationship for production Honeywell VGT Turbine

An exponential fit was used and incorporated into the Simulink heat transfer model.

Steady State Results

A summary of the steady state test conditions and the results obtained is given in Table 2.

Table 2 – Summary of test conditions and results obtained from simulation. 'Comp' is abbreviated from 'compression' and 'Exp' is abbreviated from 'Expansion'. The dT values represent the temperature change of the gas entering and leaving that section of the model – for example, at 2500RPM & 300Nm, the air temperature increased by 2.8K within the compressor housing before undergoing the compression process. Wall error values represent the % error between actual and predicted housing temperatures.

maximum compressor and turbine wall temperature errors were 2.9% and 8.1% respectively at the steady state test conditions.

Examining Table 2, negative values of pre and post expansion dT show that, at all conditions examined, there was energy flow from the exhaust gas to the turbine housing, even after the expansion process where the gas temperatures are reduced. This trend is reversed in the compressor, with energy flowing from the compressor housing to the intake air (shown by positive pre compression dT values) leading to an increase in air temperature prior to the compressions process. After the compression process the air temperature has risen leading to heat loss from the air to the housing under most conditions.

Speed RPM	Load Nm	MAF kg/hr	Oil Flow kg/s	Pre Comp dT °K	Post Comp dT °K	Comp Wall Error %	Pre Exp dT °K	Post Exp dT °K	Turbine Wall Error %
2500	300	403.8	0.00748	2.8	-0.8	-1.2	-5.4	-1.9	-3.2
2000	300	325.4	0.00156	5.3	-0.6	2.2	-8.4	-1.3	3.2
1500	305	221.9	0.00072	11.3	-1.6	1.5	-14.6	-3.8	5.9
1500	201	171.6	0.00072	11.8	0.1	2.9	-13.9	-6.3	5.6
2500	200	322.0	0.00075	5.0	-0.7	1.8	-7.0	-1.2	1.4
3000	150	325.2	0.00203	4.5	-1.2	-0.8	-5.9	-1.0	-0.9
2000	101	117.7	0.00317	14.8	-1.7	0.8	-14.5	-8.0	5.0
1000	150	83.5	0.01304	14.4	-2.9	-0.8	-16.8	-11.4	1.1
1000	100	60.6	0.01191	16.0	-1.0	-0.6	-16.7	-12.4	-0.4
1000	50	39.8	0.00159	20.1	-0.2	-0.8	-15.1	-11.7	-2.9
1500	50	57.9	0.00159	16.9	0.2	0.1	-14.1	-9.8	3.2
2000	50	89.8	0.00159	13.5	-0.9	0.4	-11.7	-6.4	6.6
2500	50	128.3	0.00159	10.5	-1.3	0.3	-10.0	-3.8	8.1
3000	50	190.1	0.00159	7.1	-1.8	-0.5	-6.9	-1.4	5.0

Figure 8 shows the absolute predicted and measured compressor and turbine wall temperatures at each of the steady state test points.

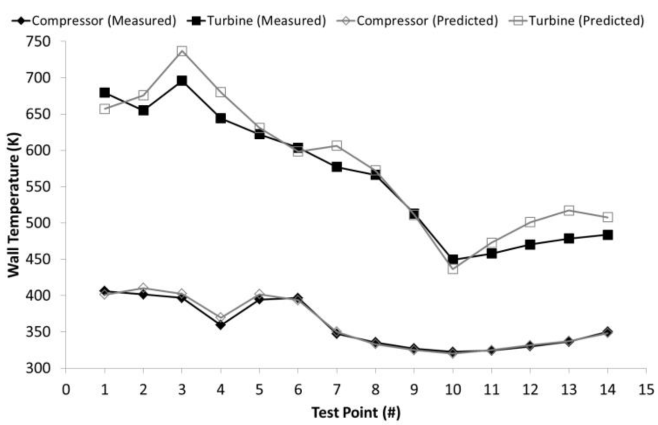

Figure 8 – Model compressor and turbine wall temperature predictions compared to measured data

It can be seen in Figure 8 that the model predictions for compressor wall temperature show very good correlation with measured data (average 0.4%, standard deviation 1.27%) and that turbine housing temperature also demonstrates good agreement (average 2.7%, standard deviation 3.58%). The

Summary/Conclusions

A simplified heat transfer model was constructed in the MatLab Simulink environment to predict thermal effects. Steady state experimental data was used to derive an empirical relationship between the Reynolds and Nusselt numbers for the examined hardware and the accuracy of the predictions were assessed under steady state conditions.

Under steady state conditions, the model was found to produce accurate predictions of compressor wall temperatures with an average error of 0.4% and standard deviation of 1.27%. Predictions of turbine housing temperature show good agreement with measured data but with a slightly higher average error of 2.7% and standard deviation of 3.58%.

Under all steady state engine operating conditions examined, heat transfer was found to be consistently from the exhaust gas to the turbine housing both before and after the expansion event resulting in a decrease in exhaust gas temperature. In contrast, within the compressor, before the compression process, heat transfer is consistently from the housing to the gas, but this trend is reversed post-compression where the gas temperature was found to be slightly higher than the wall temperature under most conditions leading to a slight reduction in gas temperature.

Transient performance is extremely important, and evaluation of the model predictions under tip-in and tip-out events was subsequently undertaken. This transient performance evaluation will be presented in an upcoming publication by the author.

This paper addresses the approach undertaken to obtain empirical data which can be used to make accurate predictions of compressor and turbine gas exit temperatures and housing temperatures. The Re/Nu relationships derived from experimental data are specific to the hardware examined which is likely to limit the transferability of the results. However, relatively few experiments are required to generate the Re/Nu relationship for new hardware which can easily be incorporated into the model described.

References

1. Shayler, P.J., Hayden, D.J. 1999. Exhaust system heat transfer and catalytic converter performance; SAE International, Paper No. 1999-01-0453, 1999.
2. Eriksson, L. 2002. Mean value models for exhaust system temperatures; SAE Paper 2002-01-0374, 2002
3. Bannister, C., Brace, C.J, Lock, G., Taylor, J., Brooks, T., Fraser, N., 2009. Experimental Characterisation of Heat Transfer in Exhaust Pipe Sections. *SAE International Journal of Materials & Manufacturing*, 1 (1), pp. 136-144
4. Bannister, C., Brace, C., Taylor, J., Brooks, T., Fraser, N., 2009. The use of multi-variate models for the prediction of heat transfer in vehicle exhaust systems. *Proceedings of the Institution of Mechanical Engineers, Part D: Journal of Automobile Engineering*, 224 (3), pp. 369-385.
5. Bannister, C.D., Brace, C.J., Taylor, J., Brooks, T. AND Fraser, N., 2011. An empirical approach to predicting heat transfer within single- and twin-skin automotive exhaust systems. *Proceedings of the Institution of Mechanical Engineers, Part D: Journal of Automobile Engineering*, 225 (7), pp. 913-929
6. Medved, T.M., Bolze, C.C., Hansen, C.E. & Barger, J.W. 1959. Heat capacity determination of mineral and synthetic engine oils, lubricants, fuels and hydraulic fluids in the temperature range 70° - 500°F. Wright Air Development Centre Technical Report, United States Air Force.

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Definitions/Abbreviations

Comp	Compression
dT	Change in Temperature (K)
Exp	Expansion
$\dot{m}$	Mass Flow (kg/hr)
MAF	Mass Air Flow (kg/hr)
Nu	Nusselt Number
P_{in}	Inlet Pressure (bar)
P_{out}	Outlet Pressure (bar)
P_{post_exp}	Post-expansion Gas Pressure (bar)
P_{pre_exp}	Pre-expansion Gas Pressure (bar)
Pr	Prandtl Number
Re	Reynolds Number
T_{in}	Inlet Temperature (K)
T_{out}	Outlet Temperature (K)
T_{post_exp}	Post-expansion Gas Temperature (K)
T_{pre_exp}	Pre-expansion Gas Temperature (K)
$T_{T\_housing}$	Turbine Housing Temperature (K)
VGT	Variable Geometry Turbocharger